NASA TECHNICAL TRANSLATION

NASA TT F-447

C.1

NASA TT F-447

ON FRICTION AND LEAKAGE LOSSES IN THE WORKING BLADE-RINGS OF RADIAL-FLOW TURBINES IN HELICOPTER AND TRANSPORT GAS-TURBINE UNITS

by N. F. Galitskiy

Translated from *Izvestiya Vysshikh Uchebnykh Zavedeniy, Seriya Aviatsionnaya Tekhnika,* Volume 7, No. 4, 1964

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • JANUARY 1967

NASA TT F-441

ON FRICTION AND LEAKAGE LOSSES IN THE WORKING BLADE-RINGS OF RADIAL-FLOW TURBINES IN HELICOPTER AND TRANSPORT GAS-TURBINE UNITS

By N. F. Galitskiy

Translation of "O poteryakh treniya i ventilyatsii v rabochikh ventsakh radial'nykh turbin vertoletnykh i transportnykh GTU." Izvestiya Vysshikh Uchebnykh Zavedeniy, Seriya Aviatsionnaya Tekhnika, Vol. 7, No. 4, 1964, pp. 60-66.

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

For sale by the Clearinghouse for Federal Scientific and Technical Information
Springfield, Virginia 22151 – Price $1.00

ON FRICTION AND LEAKAGE LOSSES IN THE WORKING BLADE-RINGS OF RADIAL-FLOW TURBINES IN HELICOPTER AND TRANSPORT GAS-TURBINE UNITS

N. F. Galitskiy

ABSTRACT

A working blade-ring containing 208 blades from a Ljungström turbine was rotated in the forward and reverse direction with variable rpm under the following conditions: in free space, with coupled turbine guide blade-rings and in the annular space between the free floating rings and the guide blade-rings. The experimental installation is described in detail.

The literature lacks studies of power losses from no-load operation of working blade-rings of radial-flow turbines. /60**

Because of the possible application of single- and double-rotor radial-flow turbines in helicopter gas-turbine units (e.g., see British Patent No. 820337, Class 4, 1956), where the degree of the losses mentioned affects the rate of helicopter descent when the gas-turbine unit is not operating and makes it necessary to install a bypass coupling (and also because of the use of these turbines in transportation power plants requiring the power turbine to go into reverse), research was conducted which should in some measure fill in this gap.

One object of the investigation was the blading taken from a multi-stage radial-flow Ljungström turbine. As is known, the blading design of Ljungström radial-flow turbines possesses high thermoelasticity and the ability to maintain the radial clearances between the blade-rings almost constant during substantial, rapid changes in operating conditions and also when the blading elements are heated to a high level. For example, an aviation, gas, two-rotor radial-flow gas-turbine using the Ljungström system operated successfully in 1935 at an initial gas temperature of 923 to 973°K (Ref. 1). Owing to these properties the Ljungström blading design is most suitable for turbines which must be highly maneuverable, reliable, and economical. It is also suitable for executing under multistage turbine conditions (in the absence of vacuum) such well-known methods of reducing the size of friction and leakage losses in working blade-rings in protracted no-load rotation as covering them with annular jackets and decreasing the density of the medium (by substantially raising the temperature of the medium, using for this purpose the heat into which friction losses are converted).

---

* Note: Numbers in the margin indicate pagination in the original foreign text.

Conditions, Means, and Methods of the Experiments

The research was conducted on a training and experimental unit whose diagram (for testing radial-flow blade-rings) is given in Figure 1a. A working blade-ring 1 from a Ljungström turbine with an outside diameter of 366.4 mm and 208 blades 7.7 mm wide and 18 mm long was fastened directly onto the shaft of a high-frequency electric motor 2, having a power of 9 kw and a speed of 8000 rpm; it was rotated forward and in reverse in an atmos- /61 pheric air medium in vessel 10 at a variable number of rpm under the following conditions:

(1) in free space with no elements of the flow complex,

(2) with guide blade-rings 8 coupled to it (Figure 1b), and

(3) in the annular space between the free-floating rings of the guide blade-rings (Figure 1c).

Axial displacement of guide blade-rings 8 from position b to c was effectuated both when the working blade-ring was rotating and was at rest, without interference between the blade-rings by using the three angle brackets 7 to turn supporting disc 9 which bears the guide blade-rings. The disc was fastened by a threaded connection to the lower cross-support of the electric motor coaxially with the motor shaft. Axial displacement of the guide blade-rings in motion was accomplished in the same way as displacement of the stress-relief discs in the first Ljungström radial-flow turbine, in the labyrinth-seals of which the clearances were considerably less than in the case in question (Ref. 2). The number of rotations was mechanically measured by tachometer 4 with an accuracy of 5%; temperature of the medium was determined by a mercury thermometer near the blade-rings with an accuracy of 0.5°; barometric pressure was taken from Weather Bureau data. The moment of forces acting on the radial blade-ring and equal to the torque at the electric motor shaft was found from the moment of reaction on the stator measured in the installation by the torque angle of torsion shaft 3, on which the stator is suspended and together with /62 which it forms a torsion balance. The torque angle was measured visually by means of the stationary scale and needle 5 fixed to the stator. The torsiometric system was calibrated on the device by means of a spring dynamometer and balance, with the electric motor running and at rest. Calibration took place before and after the experiments. The results of repeated calibrations throughout a year were mutually congruent when using both dynamometer and balance. The power lost to rotation of the radial blade-ring was determined from the formula

$$N = C \cdot \frac{\varphi \cdot L}{75} \cdot \frac{\pi \cdot n}{30} = \frac{1}{16{,}7} \cdot \frac{\varphi \cdot n}{1000}.$$

Here N is power, kw;

$C = \frac{G}{\phi} \cdot g$ - characteristic of the torsiometric system, H/mm;

G - calibration force, H;

ϕ - length of arc on scale corresponding to torque angle, mm;

L - calibrating lever arm, m;

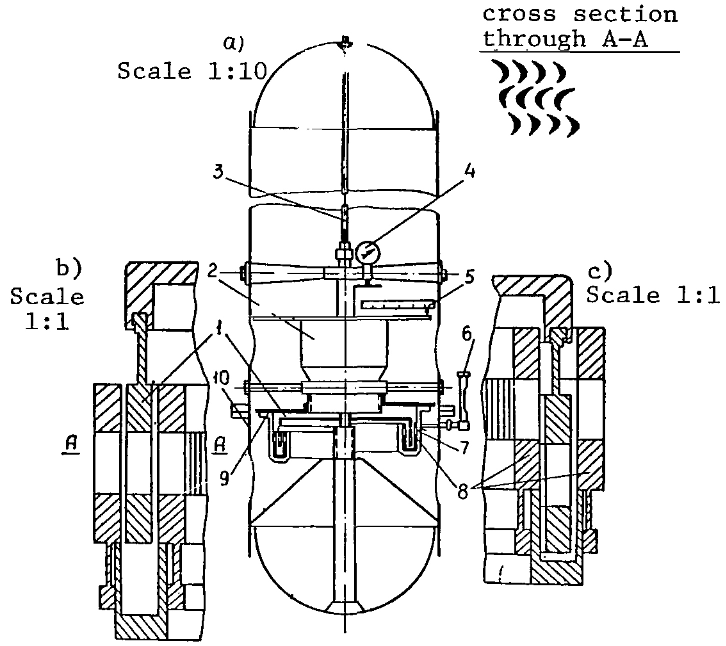

Figure 1

Diagram of Installation

(1) - Working blade-ring; (2) - Electric motor; (3) - Torsion shaft; (4) - Tachometer; (5) - Scale and needle; (6) - Thermometer; (7) - Angle bracket; (8) - Guide blade-rings; (9) - Supporting disc; (10) - Vessel.

n - rpm; and
g - acceleration of force of gravity, m/sec^2.

For plotting graphs showing the dependence of power loss with friction and of leakage on speed of rotation, the quantity $\frac{\phi \cdot n}{1000}$, proportional to the power and reduced to the conditions of air pressure 1.013 bar or 760 mm Hg and temperature 293°K, was found to be more convenient. Total error in determining power in the low rpm region was about 9%. Reduction to the above conditions was accomplished by formula

$$\left(\frac{\varphi \cdot n}{1000}\right)_{pr} = \frac{\varphi \cdot n}{1000} \cdot \frac{760}{B} \cdot \frac{T}{293},$$

where B is barometric pressure, mm Hg, and
T - temperature of the medium, °K.

Experimental Results

Figure 2 gives the results of the experiments in a logarithmic system of coordinates.

Curves 1 and 5 correspond to no-load rotation of the working blade-ring in reverse and forward directions between guide blade-rings in the position shown in Figure 1b.

Curves 2 and 4 characterize the degree of friction and the leakage losses in a single working blade-ring rotating in free space in reverse and forward directions, respectively.

Curves 3 and 6 give the degree of the above-mentioned losses in reverse and forward rotation of the working blade-ring in the annular space between the floating rings of the guide blade-rings in the position shown in Figure 1c. A comparison of the ordinates of curves 1 and 5, 2 and 4, and also 3 and 6 shows that when the direction of no-load rotation of the working blade-ring is changed from forward to reverse, other conditions remaining the same, the friction and leadkage losses are increased - in the first case (intermediate stage) 15 to 20 times, in the second (single blade-ring) 9 - 10 times, and in the third (working blade-ring in annular jacket) 2 - 2.5 times. /63

As the slope of the graphs show, in all the above cases the degree of friction and leakage loss in forward no-load rotation of the working blade-ring is proportional to the power of the number of revolutions with an exponent of 2.85 to 2.9, while in reverse rotation, the exponent is 3 and 3.3, which in the given coordinate system equals the tangent of the angle of inclination of the graphs to the abscissa axis.

If we adopt as a standard of comparison the level of losses from friction and leakage in the single working blade-ring rotating in a free medium, then the effect of the structural elements on the degree of losses mentioned in the working blade-ring may be characterized by the following relationships.

During forward rotation of the working blade-ring, as curves 4 and 5 show, the guide blade-rings decrease the friction and leakage losses by approximately a factor of 1.5. The explanation for this is the decreased flow-rate of the medium leaking through the working blade-ring with guide devices, as compared with the case of similar rotation of the working blade-ring without these devices in free space.

Bilateral covering of the blades of the working blade-ring with the free-floating rings of the guide blade-rings, as curves 4 and 6 indicate, leads to reduction of the mentioned losses in the forward rotational direction by a factor of 1.6 to 1.7, as compared with the case of working blade-ring rotation in the same direction in a free medium, and by a factor of 1.1 in comparison with forward rotation under intermediate stage conditions. The explanation of this lies also in the reduction of leakage of the medium.

In reverse no-load rotation of the working blade-ring, the presence of

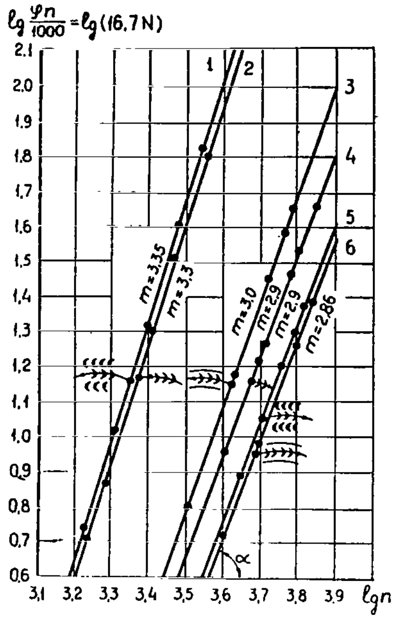

Figure 2

Results Derived from Testing Working Blade-Ring of Ljungström Radial-Flow Turbine, m = tan α

guide blade-rings, as is evident from a comparison of the ordinates of curves 1 and 2, augments friction and leakage losses by a factor of 1.1 with respect to a single blade-ring rotating in reverse in free space.

From a comparison of the ordinates of curves 2 and 3, as well as of curves 1 and 3, it follows that covering of the blades of the working blade-ring during reverse rotation with the floating rings of the guide blade-rings of the adjacent stages (which thus form an annular space) or with a special ring jacket, makes it possible to reduce friction and leakage losses 8 to 9 times, as compared to the case of reverse rotation of the single blade-ring in a free medium, and 9 to 10 times as compared to the case of similar rotation under intermediate-stage conditions. The results are in satisfactory agreement with the results derived from investigating other radial blade-rings. They also agree with the specially formulated experiments partially published in (Ref. 3, 4), making it possible to establish the quantitative relationships between the power lost due to leakage of the medium and in overcoming blade-edge friction, and thus making it possible to clarify the nature and significance of the losses arising from interaction of the blades with the medium under the conditions of free space and in annular jackets. Visual observation and measurement of the angles of flow of the medium at inlet and outlet of the radial blade-ring rotating in free space demonstrated that it interacts with the medium primarily like the working blade-ring of the simplest /64 blower of radial-flow type.

The theoretical computations, which take into consideration leakage of the medium and edge friction and which are confirmed by experimental investigations (Ref. 3), give the following formula for determining friction and leakage power in the radial-flow blade-ring:

$$N = k \cdot d^4 \cdot l \left(\frac{n}{1000}\right)^m \cdot \rho.$$

In groups of radial-flow stages with blades of identical outline, with comparatively little difference in their lengths, the latter formula may be represented as

$$N = k \cdot g l_{av} \cdot \rho_{av} \cdot \left(\frac{n}{1000}\right)^m \cdot \sum_1^z d_i^4 .$$

When there is a considerable number of uniform stages in a group with identical width in working and guide blade-rings and size of inter-blade-ring clearances, the sum of the fourth power of the external diameters of the working blade-rings may be approximated by the integration:

$$\sum_1^z d_i^4 = \sum_1^z [d_1 + 4b(z-1)]^4 \simeq \int_1^z [d_1 + 4b(z-1)]^4 dz = \frac{[d_1 + 4b(z-1)]^5 - d_1^5}{20 \cdot b},$$

$$\sum_1^z d_i^4 \simeq \frac{d_z^5 - d_1^5}{20 \cdot b} .$$

Here the following notation is used:

N - friction and leakage power, kw;
n - number of revolutions, rpm;
m - exponent for number of revolutions;
l, $l_{av} = \frac{l_1 + l_z}{2}$ - length and average length, respectively, of working blades, cm;
d - outside diameter of working blade-ring m;
ρ, ρ_{av} $\frac{\rho_1 + \rho_z}{2}$ - density and average density, respectively, of medium, kg/m^3;
b - normalized width of blades which equals the sum of blade-ring width plus radial clearance, m;
z - number of stages; and
k - coefficient allowing for effect of direction of rotation, geometrical angles, width and spacing of blades, structural elements, and size of radial clearances.

The superscripts and subscripts 1 and z distinguish the quantities referring to the first and last stages of the group. /65

According to experiments with radial-flow blade-rings, the exponent of the number of revolutions may in all cases of forward no-load rotation of the blade-ring be assumed to equal 2.85 - 2.9 (in round numbers, 2.9), while in reverse rotation it may be assumed to be 3 - 3.3 (in round numbers, 3).

Depending on circumstances, the coefficient k may be assumed to be as follows, based on the results of testing the Ljungström radial-flow blade-ring:

TABLE

	Rotation	
	Forward	Reverse
1. Blade-ring in free medium	0.257	2.57
2. Blade-ring in intermediate stage	0.162	2.94
3. Blade-ring in annular jacket or between free-floating guide blade-rings with no inter-blade-ring labyrinth seals	0.147	0.294

With inter-blade-ring labyrinth seals, in which the clearances are always less than those between the blade-rings, there is an additional 2 to 2.5-fold reduction in leakage and friction losses in the third case (Ref. 4), which is stipulated by the reduction in leakage of the medium.

The feasibility of a substantial additional reduction in leakage loss in the third case, by arranging the sealing to maintain sufficiently large clearances between the bulky rotating and stationary parts, is of great practical importance, since the reliability of the turbine rises at the same time that losses are reduced. The possible interferences of the relatively thin sealing elements during sharp changes in operating conditions cannot result in damage and significant change in friction and leakage losses, since the size of the clearances in the inter-blade-ring seals of existing radial-flow turbines is 5 to 10 times less than the clearances between the blade-rings (Ref. 5).

A restriction on the length of this article forbids a detailed discussion of the content of all experiments made. Therefore, we shall limit ourselves to listing the basic results of the work done on friction and leakage losses in radial-flow blade-rings.

1. Friction and leakage losses were studied in radial-flow blade-rings during no-load rotation in forward and reverse directions with variable rpm:

(a) in free space, (b) in annular sealed and unsealed labyrinth jackets, (c) under conditions of a single centrifugal stage, (d) under intermediate stage conditions of multistage turbines, and (e) in the annular space between free-floating rings of the guide blade-rings of the adjacent stages of multistage turbines.

2. A determination has been made of the quantitative relationship between the power lost due to leakage of the medium and to friction of the blade edges; the significance of losses during no-load rotation of radial-flow blade-rings in free space and in annular jackets has been clarified.

3. The effect of the following factors on degree of friction and leakage power loss in no-load rotation of the radial-flow blade-ring has been determined:

(a) direction of rotation, (b) number of turns (rpm), (c) length of working blades, (d) width of working blade-rings, (e) geometrical exit angle of effusor blades in a single centrigugal stage, (f) structural elements of the flow complex, and (g) inter-blade-ring labyrinth seals and radial clearances during rotation in annular jackets and in the annular space between the free-floating rings of guide blade-rings. /66

4. The amount of friction and leakage loss in radial-flow and axial-flow blade-rings has been compared experimentally.

5. Structural methods of reducing friction and leakage losses in radial-flow blade-rings have been discussed.

6. Formulas and coefficients have been suggested for practical use to facilitate the computation of the amount of power lost when radial-flow blade-rings rotate under no-load in single- and multistage radial-flow turbines.

REFERENCES

1. Lysholm, A. J. R. The Engineer, No. 4756, 4757, 1947.

2. Ljungström, F. The Institution of Mechanical Engineers Proceedings, No. 2, 160, 1949.

3. Galitskiy, N. F. Trudy Leningradskogo Korablestroitel'nogo Instituta, No. 33, 1961.

4. Idem. Ibid., No. 34, 1961.

5. Bershadskiy, M. L., and Sluchayev, M. A. Radial Flow Turbines (Radial'nyye Turbiny). Gosenergoizdat, Moscow, 1950.

6. Zhiritskiy, G. S. Aviation Gas-Turbines (Aviatsionnyye Gazovyye Turbiny). Oborongiz, Moscow, 1950.

Received April 17, 1964

Scientific Translation Service
4849 Tocaloma Lane
La Canada, California

NASA-Langley, 1967 F-447

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION **TECHNICAL TRANSLATION EVALUATION**	*Budget Bureau No. 104-R037* *Approval Expires: Sept. 30, 1969*
TO: THE USERS OF THIS TRANSLATION →	NASA TTF NO. 447

MAINTAINING THE QUALITY OF NASA TRANSLATIONS REQUIRES A CONTINUOUS EVALUATION PROGRAM. PLEASE COMPLETE AND MAIL THIS FORM TO AID IN THE EVALUATION OF THE USEFULNESS AND QUALITY OF THE TRANSLATING SERVICE.

THIS PUBLICATION *(Check one or more)*

- [ ] FURNISHED VALUABLE NEW DATA OR A NEW APPROACH TO RESEARCH.
- [ ] VERIFIED INFORMATION AVAILABLE FROM OTHER SOURCES.
- [ ] FURNISHED INTERESTING BACKGROUND INFORMATION.
- [ ] OTHER *(Explain)*: ______________________

FOLD LINE **FOLD LINE**

TRANSLATION TEXT *(Check one)*

- [ ] IS TECHNICALLY ACCURATE.
- [ ] IS SUFFICIENTLY ACCURATE FOR OUR PURPOSE.
- [ ] IS SATISFACTORY, BUT CONTAINS MINOR ERRORS.
- [ ] IS UNSATISFACTORY BECAUSE OF *(Check one or more)*:
  - [ ] POOR TERMINOLOGY.
  - [ ] NUMERICAL INACCURACIES.
  - [ ] INCOMPLETE TRANSLATION.
  - [ ] ILLEGIBLE SYMBOLS, TABULATIONS, OR CURVES.
  - [ ] OTHER *(Explain)*: ______________________

FOLD LINE **FOLD LINE**

REMARKS

FROM	DATE

NOTE: REMOVE THIS SHEET FROM THE PUBLICATION, FOLD AS INDICATED, STAPLE OR TAPE, AND MAIL. NO POSTAGE NECESSARY.

NASA Form 1332 AUG 66

GPO 915-777

CUT ALONG THIS LINE

CUT ALONG THIS LINE

"The aeronautical and space activities of the United States shall be conducted so as to contribute . . . to the expansion of human knowledge of phenomena in the atmosphere and space. The Administration shall provide for the widest practicable and appropriate dissemination of information concerning its activities and the results thereof."

—NATIONAL AERONAUTICS AND SPACE ACT OF 1958

NASA SCIENTIFIC AND TECHNICAL PUBLICATIONS

TECHNICAL REPORTS: Scientific and technical information considered important, complete, and a lasting contribution to existing knowledge.

TECHNICAL NOTES: Information less broad in scope but nevertheless of importance as a contribution to existing knowledge.

TECHNICAL MEMORANDUMS: Information receiving limited distribution because of preliminary data, security classification, or other reasons.

CONTRACTOR REPORTS: Technical information generated in connection with a NASA contract or grant and released under NASA auspices.

TECHNICAL TRANSLATIONS: Information published in a foreign language considered to merit NASA distribution in English.

TECHNICAL REPRINTS: Information derived from NASA activities and initially published in the form of journal articles.

SPECIAL PUBLICATIONS: Information derived from or of value to NASA activities but not necessarily reporting the results of individual NASA-programmed scientific efforts. Publications include conference proceedings, monographs, data compilations, handbooks, sourcebooks, and special bibliographies.

Details on the availability of these publications may be obtained from:

SCIENTIFIC AND TECHNICAL INFORMATION DIVISION

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

Washington, D.C. 20546